

Confirmation of Large-Periphery Compressible Gas Flow Model for Microvalves

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ABSTRACT

Previously, a compressible gas flow model was proposed, which couples microvalve structural parameters to gas parameters and flow boundary conditions, and which explained the behavior of compressible flow over a large range of conditions. However, only a limited set of data, from a single orifice for the valve seat, was used to substantiate the model. Multiple-hole or large-periphery valve seat structures were not measured. In this work, the proposed comprehensive compressible gas flow model for microvalves is confirmed using measurements of flow through multi-hole valve seat structures in microvalves.

1. INTRODUCTION

Microvalves are used in a wide variety of application areas. Control of gas flow makes them attractive for: semiconductor process gas control and distribution; automotive manifold air pressure, mass air flow, and anti-lock brake control; household natural gas control; and a wide variety of other applications. The tremendous, contemporary interest in arrays of microvalves and related microflow-control components also demands an improved model -- especially for use in the compact modeling of microflow systems, similar to the modeling constructs employed in the design and analysis of microelectronic systems [1].

While microvalves have been the subject of research and development for some time, flow modeling efforts have lagged behind, especially in the realm of compressible gas flow. Finite element analysis (FEA) can, of course, be used at any time. However, FEA is always slow and cumbersome, and in any case cannot provide the design framework either to optimize a single microvalve as a function of manufacturing parameters and boundary conditions, or to simulate a network of microvalves and other microflow devices. Such a design framework exists for integrated circuits, using SPICE and its derivatives. There is a need for a similar, compact-model framework for microflow devices, which can accommodate steady-state and transient analysis of compressible flow. The model confirmed herein goes far in filling this need. It connects microvalve flow to all relevant manufacturing parameters. And, it connects microvalve flow to the thermodynamics (temperature, pressure, and gas type boundary conditions) of compressible flow.

Other flow models related to microvalves have been reported. A loss-coefficient-based model which accounts for structural parameters was proposed, and compared to FEA data of air flow in a microvalve [2]. A model for an integrated gas flow controller, covering the Knudsen or transition regime of flow, has been described [3]. A compact flow model for fluidic networks has been described, which does not, however, include valves [4]. FEA techniques have been used [5]. A flow model for a pneumatically actuated microvalve, which includes effects of channel pressure drops on the membrane actuation, has been developed [6]. However, Ref. [6] assumes the flow is incompressible, even in the case of gases, which limits its applicability.

In our own work, modeling began with a loss-coefficient-based model [7]. This model was unsatisfactory, in that it had four fitting parameters. In addition, these parameters required re-characterization for any change in structure, or in temperature or pressure boundary condition. Subsequently, a satisfactory model for microvalves having a single and simple-structured valve seat was created [8]. This model utilized a characterization technique which led to a constant coefficient of discharge, across a wide range of structures, pressures, and flows. However, this model was limited, in that multiple valve seats, or seats with complex structure, could not be accommodated [9]. As a consequence, the model described in [10], which is confirmed here, became necessary.

The results detailed below demonstrate the efficacy of the model. However, they also indicate subtleties which must be considered in valve flow design, including: maximizing flow conductance (minimizing pressure loss) in the channels leading up to, and away from, the valve seat; and accounting for parasitic orifices in the valve inlet.

2. THEORY

Figure 1 shows the equations, which constitute the model as described in [10]. A complete derivation of the model equations will appear elsewhere [11]. In essence, the model assumes that microvalve flow can be treated through the combination of the theoretical, isentropic, adiabatic flow of an ideal gas through an orifice, with a model for the effective area of that orifice.

C_d is the coefficient of discharge. C_v accounts for departures of the flow from an ideal orifice. For the microvalves studied here, this parameter is taken to be unity. α and δ are gas-dependent functions of the specific heat ratio γ . R is the universal gas constant, divided by the gas-dependent molecular weight. T is the stagnant temperature associated with the gas at the inlet to the microvalve. P_{in} and P_{out} are the inlet and outlet pressure boundary conditions, respectively. The gap ratio parameter r defines the ratio between the gap z , and the hydraulic diameter D_h . (The hydraulic diameter is defined as $D=4A/W$, where W is the length of the periphery of the valve seat.) This ratio delineates the flow, as a function of r , into three regimes: seat-controlled flow; transition flow; and orifice-controlled flow [9]. For ratios of $r=z/D_h < r_0=0.15$ (seat-controlled region), the measured flow is found to be linear in r . For large values of r (orifice-controlled region), the effective area approaches the actual area A asymptotically.

$\dot{m}_{sonic} = C_d C_v A_{eff} \alpha(\gamma) \frac{P_{in}}{\sqrt{RT}}$ $\dot{m}_{subsonic} = C_d C_v A_{eff} \delta(\gamma) \frac{P_{in}}{\sqrt{RT}} \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma+1}{2\gamma}} \sqrt{\left(\frac{P_{in}}{P_{out}}\right)^{\frac{\gamma-1}{\gamma}} - 1}$	$A_{eff} = Wz = WD_h r \quad \left(0 < \frac{z}{D_h} < r_0\right)$ $A_{eff} = WD_h r_0 + (A - WD_h r_0) \left[1 - \exp\left(-\frac{r - r_0}{\eta}\right)\right] \quad \left(r_0 < \frac{z}{D_h}\right)$ $A_{eff} = A \quad \left(\frac{z}{D_h} \gg 1\right)$
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Figure 1: Model equations for microvalve flow.

The left side of Figure 1 shows the theoretical flow of a compressible, ideal gas through an orifice. All thermodynamic conditions (gas type, temperature) and boundary conditions (pressure, temperature) are accounted for. The right side of Figure 1 shows the model parameters for the effective area of the orifice, which constitutes the microvalve. The model for the effective area accounts for all of the structural parameters involved in the microvalve, which are essential to its design and analysis.

3. MEASUREMENTS

Figure 2 shows two versions of the valve seat arrays measured in this work. Figure 2a shows a simulation of a 7-hole valve seat array, using the AnisE simulation from IntelliSense. Figure 2b shows a photomicrograph of an actual 23-hole valve seat array.

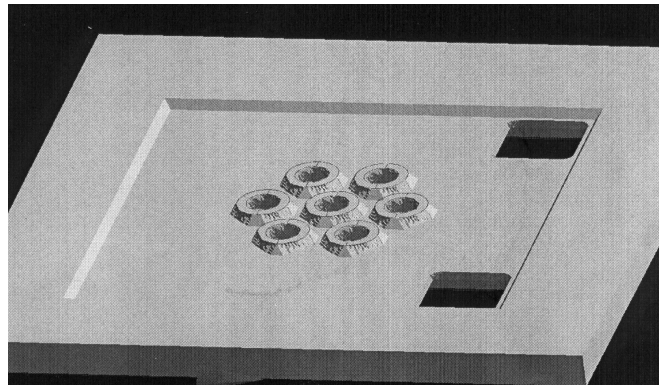


Figure 2b: AnisE Simulation of 7-hole valve seat array

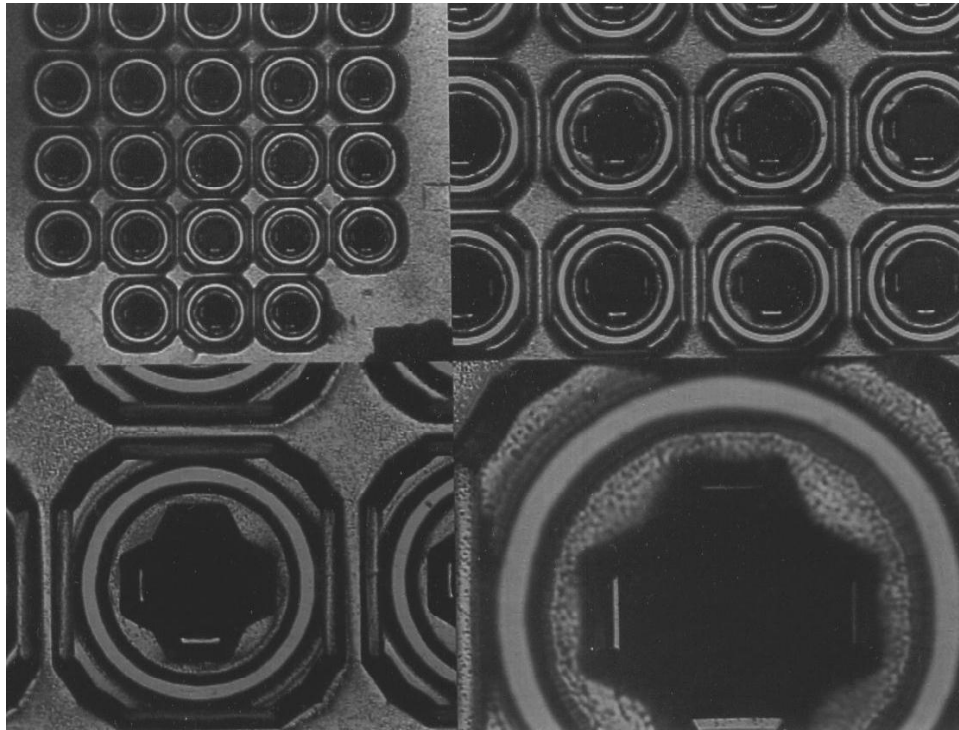


Figure 2b: Photomicrographs of 23-hole valve seat array.

Figure 3 shows a schematic cross-section, depicting the parasitic orifice, beneath the valve seat plane, which must be accounted for.

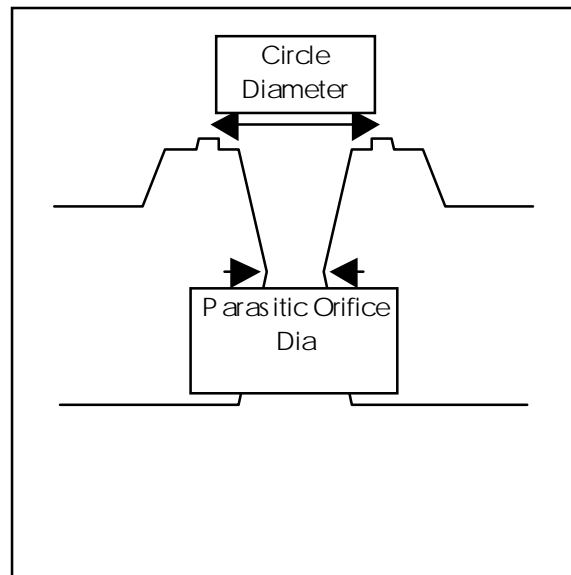


Figure 3: Cross-section of valve seat, and parasitic flow resistance (an orifice beneath the valve seat surface), which is evident from Figure 4.

Figures 4 through 7 show measurements for 5-hole, 12-hole, and 23-hole valve seat arrays, at various pressure drops. The gas is N₂. Pressure and temperature boundary conditions are indicated. The results demonstrate that the model provides a quantitative basis for microvalve seat design, where high flows are to be achieved using devices with a high periphery-to-area ratio for the valve seat.

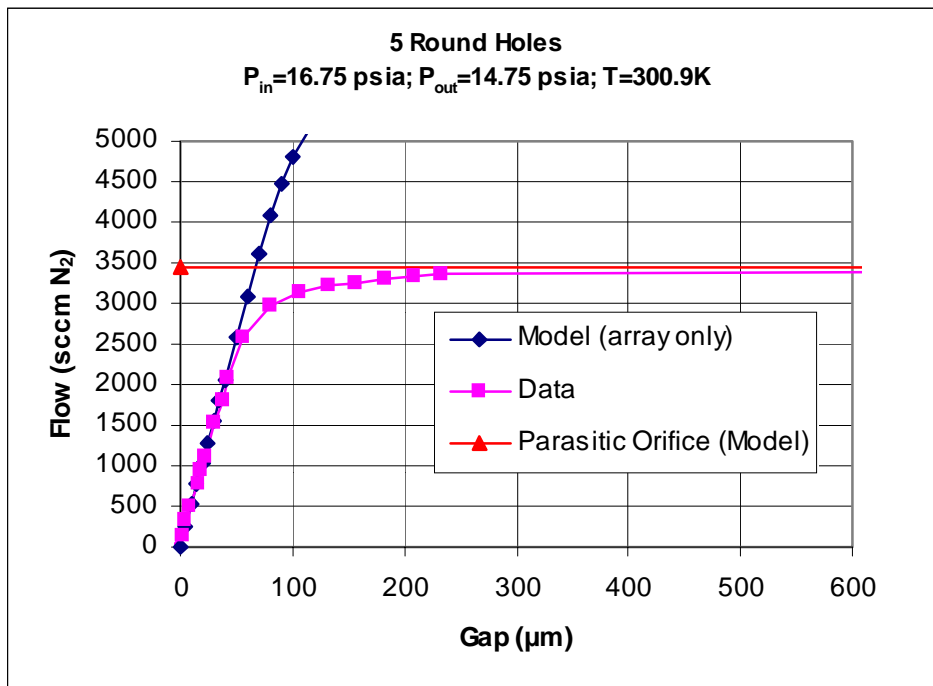


Figure 4: Measurements and models for a 5-hole valve seat array, including effect of parasitic orifice.

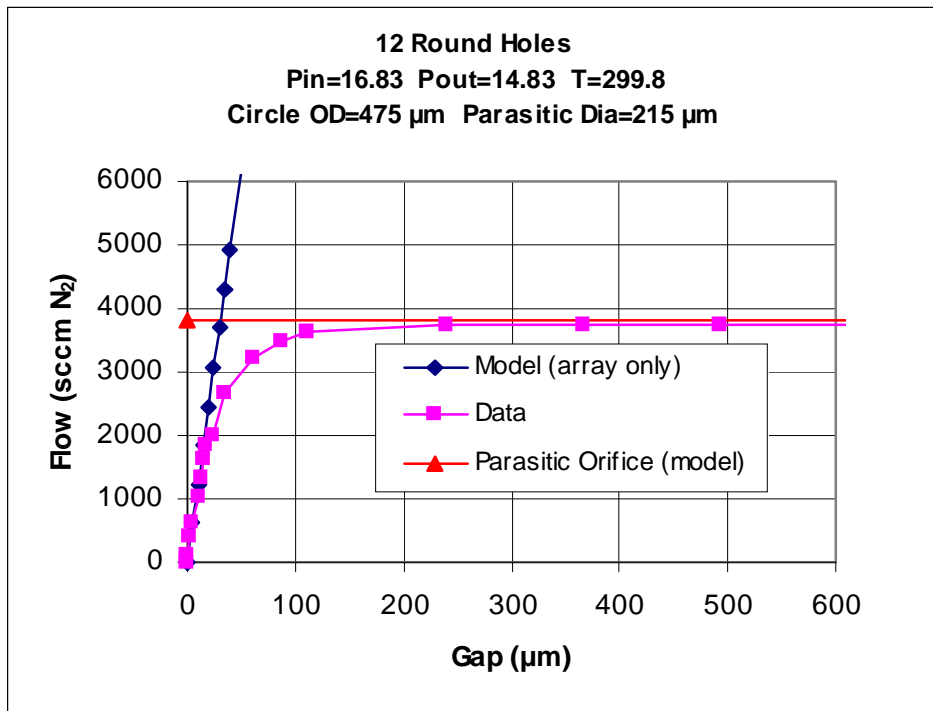


Figure 5: Measurements and models for a 12-hole valve seat array at 2 psid, including effect of parasitic orifice.

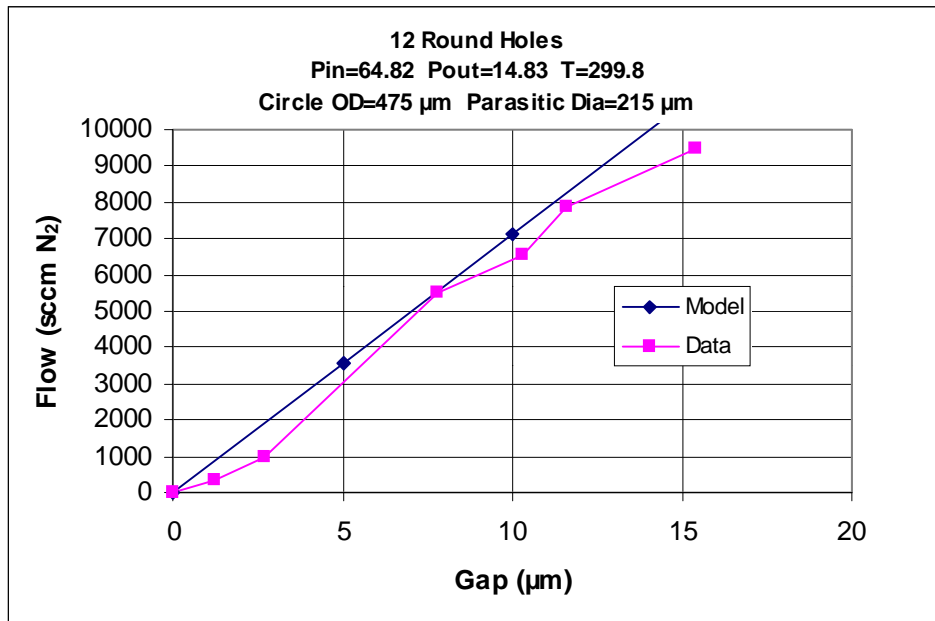


Figure 6: Measurements and models for a 12-hole valve seat array at 50 psid, including effect of parasitic orifice.

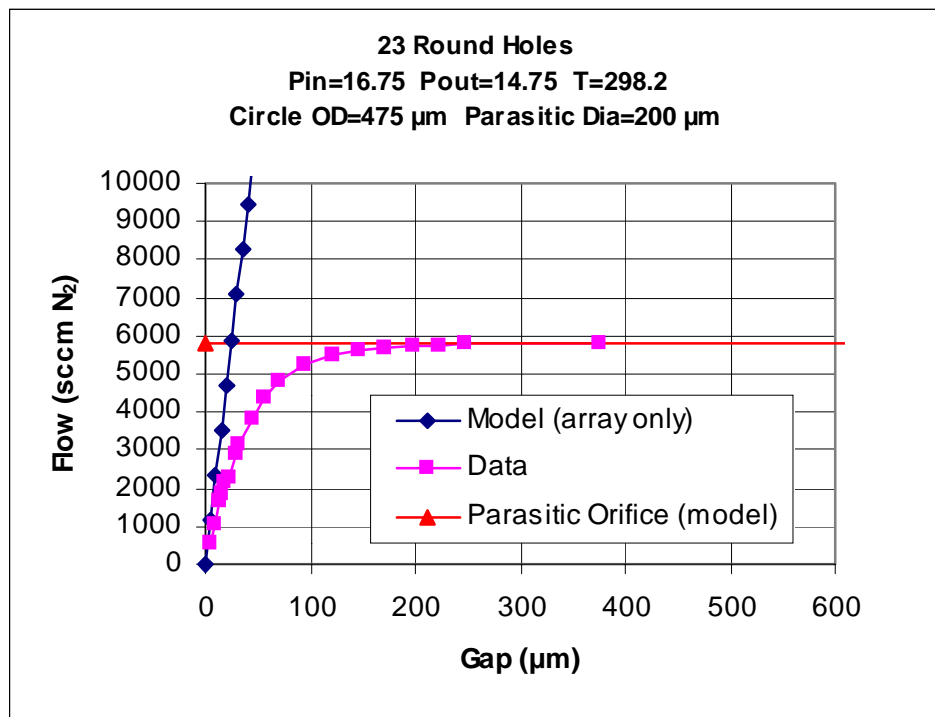


Figure 7: Measurements and models for a 23-hole valve seat array at 2 psid, including effect of parasitic orifice.

3. DISCUSSION

As is evident from Figures 4 through 7, the model correctly predicts microvalve flow behavior (provided the parasitic, in-series orifice beneath the surface of the plane of the valve seat is accounted for). This behavior is reminiscent of a MOSFET [12], provided that the gap ratio r is taken to be analogous to, or proportional to, the gate voltage in a MOSFET; and provided the microvalve inlet and outlet pressures are analogous to the MOSFET drain and source voltages, respectively. Quantitative relation of the gap r to a pneumatic input actuation is accomplished easily using standard membrane theory [13].

A surprising result of the modeling work is the very high flows which are available from very small structures. An example is shown in Figure 8. In the Figure, four different iso-flow curves are plotted. An iso-flow curve represents the combination of valve seat periphery length, and valve seat-to-membrane gap, which deliver the flow indicated in the legend, for the pressure and temperature boundary conditions given in the figure caption. As can be seen, packing more periphery length into a given area results in substantially higher flow for a given value of the gap parameter. Since valve speed is directly proportional to the gap parameter, faster response can be obtained, simply by trading periphery length against the gap parameter.

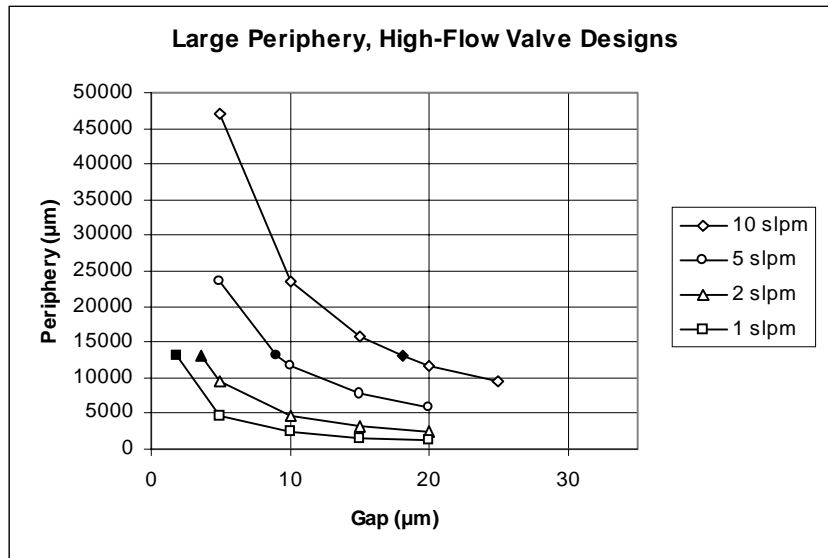


Figure 8: Modeled iso-flow curves for high flow microvalve designs. Flow conditions: N_2 ; $P_{in}=35$ psia; $P_{out}=15$ psia; $T=298K$; $C_d=0.84$. The dark symbols represent designs which fit within the areal size constraints of a present high-flow design, yet deliver three times the flow for similar boundary and ambient conditions.

With respect to the coefficient of discharge C_d , a consistent value of 0.86 has been used in the style of microvalves depicted in, for instance, Figures 2 and 3. However, theory and measurement allow more complex attributes to be incorporated into C_d , should they become necessary or desirable. An empirical dependence of C_d on pressure and temperature has been described for an orifice [14, pg. 365], which could be applied to microvalves. Also, the behavior of C_d with respect to thermodynamic parameters as well as orifice structure has been explored [15], leading to a simple relationship, amenable to compact modeling. The equation below shows the best-fit results for the measurements of Ref. [15] using a knife-edge orifice, for air flow over the temperature range 295-700 K. For pressures in the range of interest of many microvalves, this expression yields a value of C_d of approximately 0.84.

$$C_d = 0.12 \left(\frac{P_{in} - P_{out}}{P_{crit} - P_{out}} \right) + 0.626$$

Transition, or Knudsen, flow leads to departures from the model presented herein. Generally speaking, the reduction in scattering and increase in mean free path, which attend Knudsen flow, result in an increase in flow, compared to the flow expected from inspection of the predictions of a high-flow model [16]. The exact amount of increase has been examined in at least one microvalve instance [3]. Whether the conclusions reached in that effort can be extended to the work presented here remains as a topic for further exploration.

The model assumes zero heat transfer between the gas and the microvalve structure. When the flow through the structure is sonic, considerable expansion of the gas can occur, downstream from the valve seat. Such expansion is accompanied by a decrease in gas temperature. The flow model relies on the knowledge of the stagnant gas temperature, upstream of the valve seat. Any thermal coupling, direct or indirect, between the cold, expanded, downstream gas, and the warmer, stagnant, upstream gas, will call the assumption of adiabatic flow into question.

The model assumes ideal gas behavior. For gases such as helium or nitrogen, this assumption is well-founded, except at extremes of temperature or pressure which are in any case beyond the practical limit of microvalves. For other gases of interest in the manufacture of semiconductor devices, however, the assumption is less well-founded. For instance, in the case of WF_6 , the compressibility $z=p/\rho RT$ is 0.755 at a pressure of 0.6 MPa (~ 90 psia) – meaning the mass flow is expected to decrease considerably with respect to an ideal gas, where z is unity. Also, the ratio of the specific heats, γ , takes on a pressure and temperature dependence, again leading to substantial departures in flow, compared to expectations based on Equations (10). For further details on the departure of various gases from ideal behavior, the reader is referred to the work of the US National Institute for Standards and Technology [17].

4. CONCLUSIONS

Through characterization of multi-hole valve seat arrays, an improved, comprehensive microvalve gas flow model for compressible gas flow has been confirmed. It provides a powerful tool for microvalve design, for small and large flows, for microvalves of a wide range of dimensions. The model is appropriate for microvalves using any method of actuation. Because of the lumped element nature of the model -- and its relationship to parameters of interest to designers, and of analysis and control by device engineers -- it can be used to model individual devices, or arrays of devices. Its simple nature lends itself well to the further development of compact, system-level flow models for any microscale flow system.

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